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## Analysis of Food Freezing by Slot Jet Impingement

M. Jafari and P. Alavi

Faculty of Mechanical Engineering, University of Tabriz, Tabriz 51666-14766, Iran

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**Abstract:** In this study, numerical simulation using Computational Fluid Dynamics (CFD) was conducted to predict the effect of various parameters on freezing time of slab shaped food by a slot air impingement. Continuity, momentum and energy equations were solved using Fluent 6.0 a commercial Computational Fluid Dynamics (CFD) solver. In order to model the turbulent air flow, the standard Reynolds Stress Model (RSM) was applied. The results of this study show that the freezing air temperature and velocity, nozzle to food spacing are important factors affecting freezing time of slab shaped foods. This study provides useful information for optimum design and operation of food freezers.

**Key words:** Numerical simulation, freezing time, stagnation point, turbulence model, heat transfer

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### INTRODUCTION

Freezing is a widespread preservation method in food industry. The quality of frozen food is strongly dependent on freezing time. The freezing time is important in the freezing system selection for optimum food quality according to Rizvi and Mittal (1992).

Freezing time of foods can be determined experimentally or predicted approximately by either analytical or numerical methods. Freezing time determination using experimental procedures is probably the most widely used method because of its accuracy and providing valuable data on the system performance but are often too expensive, time consuming and may lack a generalized theoretical description of the process. By comparison, numerical solution method is more effective to analyze the actual situation. For freezing process modeling, many parameters representing food properties and freezing process required to be measured or collected. These parameters are shape, dimensions, water content and specific heat for unfrozen and frozen, thermal conductivity, density, initial and final temperature of the food, latent heat of fusion, conductivity of packing material, convective heat transfer coefficient, initial freezing temperature and freezing medium temperature, etc. As a result, a complete analytical solution for predicting food freezing times requires tedious calculations. Various simplified models for predicting the freezing time have been proposed by many researchers like Ramaswamy and Sablani (1996). From a physical point of view, food can be considered as a combination of a solid matrix, an aqueous phase and a gaseous phase (air and water vapor). For the freezing process, the food

can be divided into three zones: unfrozen, frozen and dehydrated. A complete mathematical model has to solve the heat transfer (freezing) and the mass transfer (weight loss) simultaneously. The mathematical model is a system of partial differential equations of the parabolic type with moving boundaries. Difficulties are encountered in the analytical solution of these equations due to the nonlinearity of moving boundary conditions. Exact solutions are possible in similarity form only in limited cases. Computational Fluid Dynamics (CFD) is a powerful numerical tool that is becoming widely used to simulate many processes in the food industry. Recent progression in computing efficacy coupled with reduced costs of CFD software packages has advanced CFD as a viable technique to provide effective and efficient design solutions. Therefore a CFD method has to be used to solve practical problems. In industrial freezing systems, an important goal is increasing heat transfer rate to reduce freezing time of foods as quickly as possible. As air impingement technology utilizes high velocity jets of air directed perpendicularly at the surface of a product to reduce thermal boundary layers, increase convective heat transfer coefficients and shorten baking, toasting, drying and freezing times, these systems have been used in food processing operations such as drying, baking and freezing as suggested by Midden (1995), LujanAcosta *et al.* (1997), Borquez *et al.* (1999), Moreira (2001) and Nitin and Karwe (2001). Due to the characteristics of fluid flow in air impingement systems this process produces a high (but also a spatially variable) convective heat transfer coefficient ( $h$ ) value as suggested by Ovadia and Walker (1998). Erdogdu *et al.* (2005) predicted the transient time-temperature change

during cooling of slab shaped geometries under impinged air conditions and investigated the spatial variation of h-value over the surface. Salvador and Mascheroni (2002) studied numerically heat and mass transfer that occur in impingement freezers. Mittald and Zhang (2000) developed an artificial network to predict the freezing time of food products of any shape. Huan *et al.* (2003) used Galerkin finite element numerical model to predict the freezing time for any shape foods at different freezing conditions. In the experimental study carried out by Sarkar and Singh (2004) a visualization technique was used to determine the importance of factors affecting efficiency of impingement systems. Wang *et al.* (2007) developed a one-dimensional mathematical model to simulate freezing time of individual food in freezing process. Soto and Borquez (2001) used a special impingement jet equipment to determine experimentally freezing time of foods. Campanone *et al.* (2005) developed a numerical model to predict weight loss of unpackaged foods during freezing. Dagtekin and Octopi (2007) numerically investigated heat transfer due to double impinging vertical slot jets onto an isothermal wall for laminar flow regime. In this study the effect of the jet Reynolds number, the jet-isothermal bottom wall spacing and the distance between two jets on heat transfer and flow field was examined. These reviews indicate that there is little research on CFD modeling of slot-jet impingement freezing of slab shaped foods. Therefore, the objective of this research was to predict the effect of various parameters on freezing time of slab shaped food by a slot air impingement using CFD solver FLUENT 6.0. In this study the slot jet is assumed to be large in the span wise direction. Hence the flow configuration can be considered as a two dimensional planar jet.

**MATERIALS AND METHODS**

**Governing equations:** The fluid flow and heat transfer are governed by three, equations: the continuity, the conservation of momentum and the conservation of energy.

- The continuity equation,

$$\frac{\partial U_j}{\partial X_j} = 0 \tag{1}$$

- The conservation of momentum (in x, y and z directions),

$$\frac{\partial U_i}{\partial t} + \frac{\partial U_i U_j}{\partial X_j} = -\frac{1}{\rho} \cdot \frac{\partial P}{\partial X_i} + \frac{\partial}{\partial X_j} \left[ \gamma \left( \frac{\partial U_i}{\partial X_j} + \frac{\partial U_j}{\partial X_i} - \langle u'_i u'_j \rangle \right) \right] \tag{2}$$

- And the conservation of energy,

$$\rho \cdot c_p \cdot \frac{\partial T}{\partial t} + \rho \cdot U_j \cdot c_p \cdot \frac{\partial T}{\partial X_j} = \frac{\partial}{\partial X_i} \left[ K \cdot \frac{\partial T}{\partial X_i} - \rho \cdot c_p \cdot \langle u'_i T' \rangle \right] + q_v \tag{3}$$

where, U is average velocity, u' is turbulent component of velocity,  $\langle u'_i u'_j \rangle$  is the average value of the fluctuating component of the velocity, T is average temperature, T' is fluctuating component of temperature, t is time, P is pressure, ρ is density, γ is kinematics viscosity of the air, K is thermal conductivity, C<sub>p</sub>, C<sub>v</sub> and q<sub>v</sub> are the heat capacity at constant pressure and volume and internal source term due freezing, respectively. In turbulent flow, the velocity magnitude fluctuates with time and these fluctuations are known as the turbulence where the velocity in the turbulent flow can be divided into the average and turbulent components. The decomposition of the flow field into the average and turbulent (fluctuating) components has isolated the effects of the fluctuations on the average flow. However, the addition of the turbulence in the Nerviier-Stokes equations, as seen above, results in additional terms, known as Reynolds stresses, leading to a closure problem increasing the number of unknowns to be solved. In order to solve this problem, a mathematical path for the calculation of the turbulence quantities must be provided. There are special turbulence models to solve this issue, e.g., k-ε, k-ω, Reynolds Stress Model (RSM) and many others. In this study the Reynolds Stress Model (RSM) with standard wall functions was applied in Fluent 6.0. The SIMPLE algorithm is used to solve the coupled system of governing equations.

**Initial and boundary conditions:** We assume that there is no mass transfer and evaporation during the freezing process although there are phase change and corresponding heat release and the slab shaped food is initially in thermal equilibrium (T = T<sub>i</sub>). Heat transfer due to radiation from food surface was neglected. With reference to Fig. 1, the mass, momentum and thermal boundary conditions are as follows:

Slot nozzle outlet: U<sub>j</sub> = constant, T = T<sub>j</sub>

Outflow boundary (EF):  $\frac{\partial U}{\partial X} = \frac{\partial V}{\partial X} = 0$

Upper boundary (AF): U = V = 0 and  $\frac{\partial T}{\partial Y} = 0$

On the food surface (BE): U = V = 0 and  $h(T - T_j) = -k \frac{\partial T}{\partial X}$

On the insulation surface (CD):  $\frac{\partial T}{\partial Y} = 0$

where, U<sub>j</sub> is velocity of air at nozzle exit, T<sub>j</sub> is temperature of freezing air, h is convective heat transfer coefficient,



Table 1: Thermal properties used in model

$C_u$ (kJ kg <sup>-1</sup> K <sup>-1</sup> )	$C_s$ (kJ kg <sup>-1</sup> K <sup>-1</sup> )	$\rho$ (kg m <sup>-3</sup> )	$L_f$ (kJ kg <sup>-1</sup> )	$k_u$ (W m <sup>-1</sup> K <sup>-1</sup> )	$k_s$ (W m <sup>-1</sup> K <sup>-1</sup> )	$T_f$ (°C)	$T_i$ (°C)
3.644	2.159	1070	188.0	0.51	1.5	-0.5	10

Here  $C_u$  is specific heat of unfrozen food,  $C_s$  is specific heat of frozen food,  $L_f$  latent heat of freezing,  $k_u$  is thermal conductivity of unfrozen food,  $k_s$  is thermal conductivity of frozen food,  $T_i$  is initial temperature of freezing food,  $T_f$  is freezing temperature of food

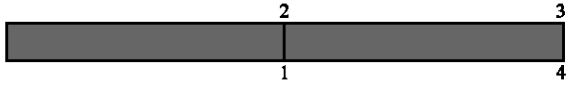


Fig. 2: Position of various points inside the food model

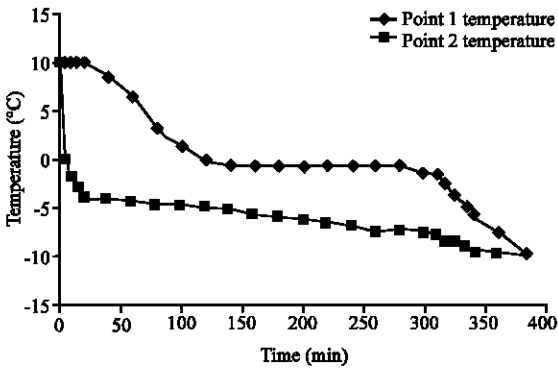


Fig. 3: Simulated temperature profiles of points 1 and 2 during freezing of the food

variation of heat transfer over the food surface for the given conditions. As observed in this figure maximum h-values were obtained at the stagnation point where the air impinges on the food surface.

**Effect of freezing air temperature on freezing time:**

Figure 5 shows the effect of freezing air temperature on freezing time. It is clear that with lower air temperature the freezing time decreases, but beyond critical point, it changes slowly. Freezing air temperature is the most important parameter in freezing and frozen storage. For freezing, when air temperature lowers, the process is accelerated, since the driving force for energy transfer increases almost linearly with the temperature difference between the surroundings and the food surface. As freezing time lowers, the same happens to mass flow by sublimation, because the time the food surface is exposed to dehydration is also lower and the surface temperature diminishes more rapidly. Figure 6 shows the effect of freezing air temperature on air domain temperature.

**Effect of freezing air velocity on freezing time:**

Figure 7 shows the effect of freezing air velocity on freezing time. It is clear that with increasing air velocity the freezing time decreases, but beyond critical point, it changes slowly. Figure 8 shows the effect of impinging air velocity

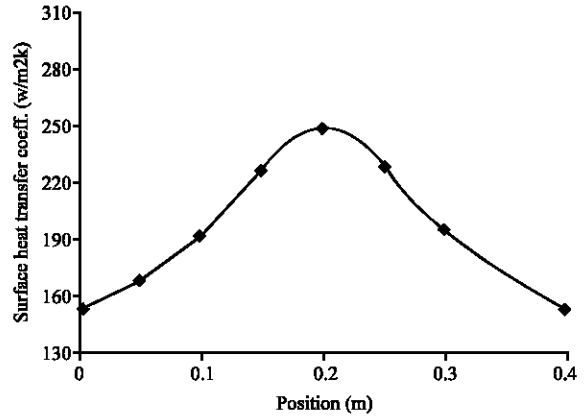


Fig. 4: Spatial variation of h-value over the food surface for the  $H = 36$  cm,  $U_j = 10$  m sec<sup>-1</sup>,  $T_j = -10^\circ\text{C}$

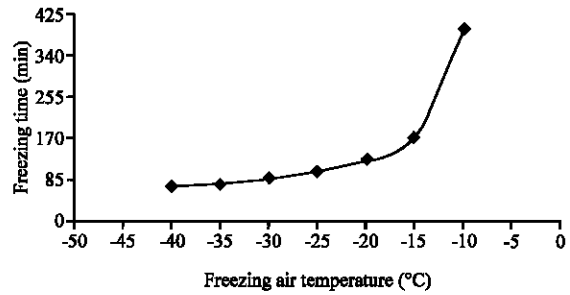


Fig. 5: Effect of freezing air temperature on freezing time

on h-value. It shows that h-value increases with impinging air velocity. This is due that flow of high velocity creates thin boundary layers, which creates a high rate of heat transfer. Figure 9 shows effect of freezing air velocity on flow and velocity field for the various air velocities. This figures show, with increasing air velocity potential core length increases. When the potential core is not completely decayed, the flow at stagnation may be laminar. The laminar flow becomes turbulent downstream, resulting in peak heat-transfer coefficients where the flow transitions from laminar to turbulent. This result is because heat transfer in turbulent flows is higher than laminar flows. When the food (impinging surface) is positioned near of the potential core of the jet, turbulence level in the jet flow approaching the food is becoming high due to an increased entrainment of surrounding air to the jet flow, therefore a

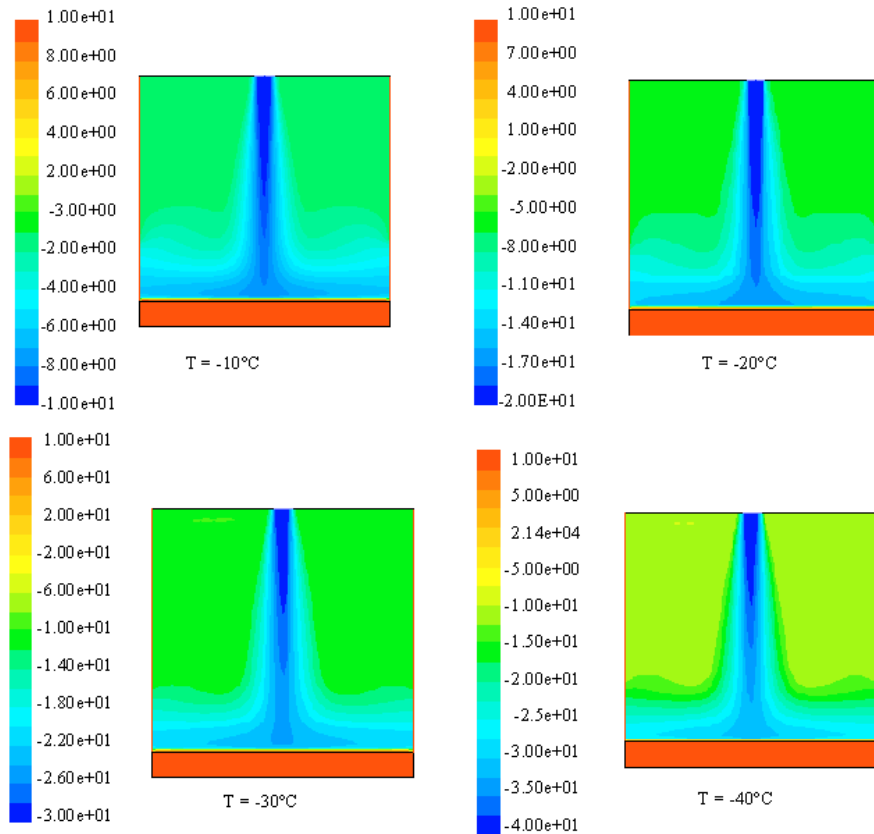


Fig. 6: Effect of freezing air temperature on temperature distribution ( $^{\circ}\text{C}$ ) in the air domain for the  $H = 36\text{ cm}$ ,  $U_j = 10\text{ m sec}^{-1}$

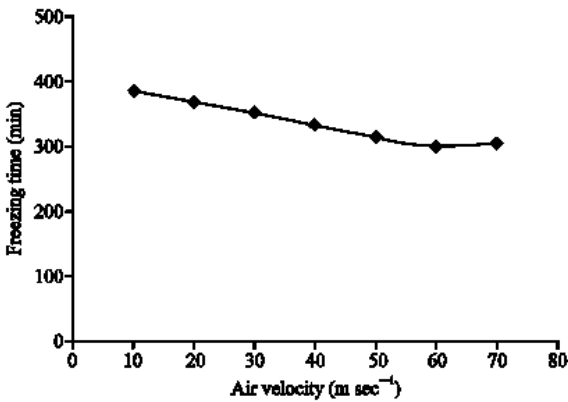


Fig. 7: Effect of freezing air velocity on freezing time

higher h-value number at the stagnation point can be expected. When the food is positioned downstream of the potential core the freezing air velocity is degraded from its nozzle exit value, the energy of the jet decayed because of the turbulence in free jet region causing lower heat-transfer coefficient.

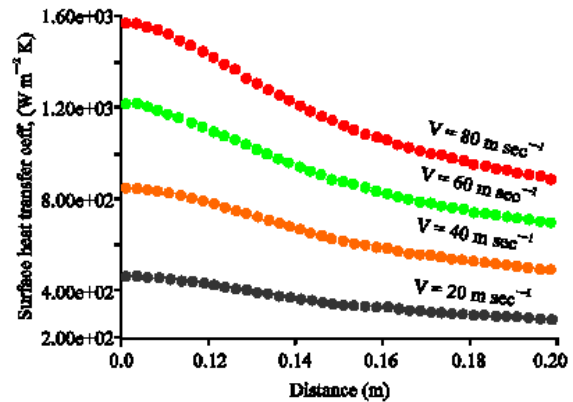


Fig. 8: Effect of freezing air velocity on h-value for  $H = 36\text{ cm}$ ,  $T_f = -10^{\circ}\text{C}$

**Effect of nozzle to food distance on freezing time:** The results of our study showed that as distance between nozzle exit and food surface decreases, freezing time increases. Figure 10 and 11 show influence of nozzle to food distance on flow pattern and on freezing time,

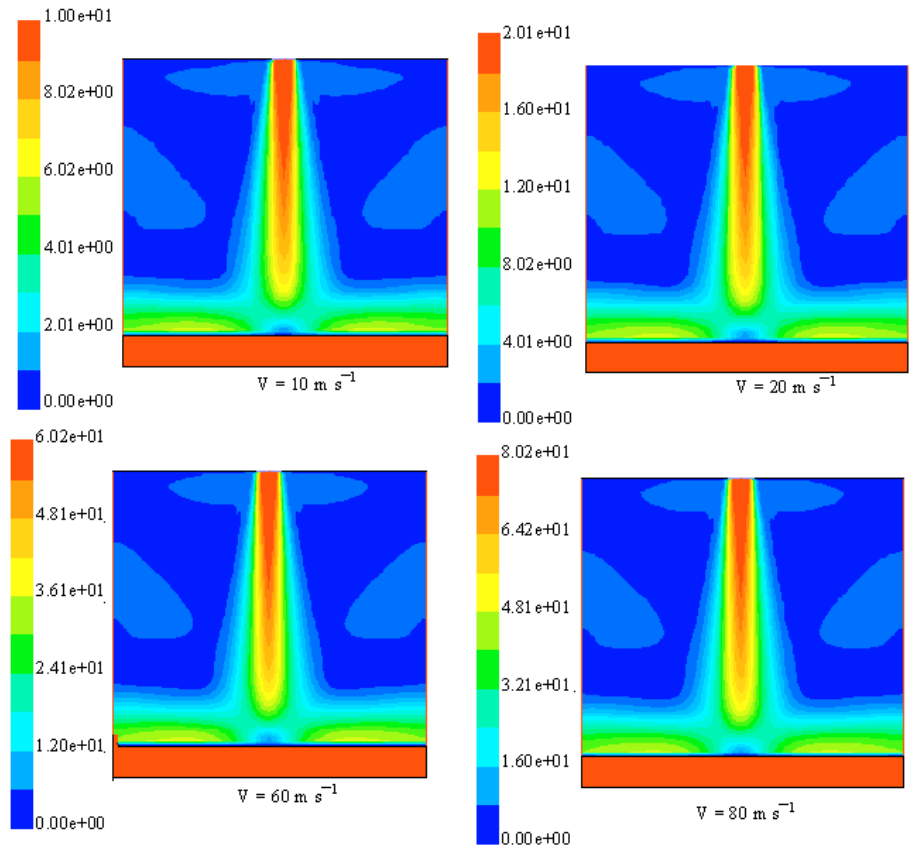


Fig. 9: Effect of freezing air velocity on flow and velocity field for the  $H = 36$  cm,  $T_j = -10^\circ\text{C}$

respectively. Contours of velocity show that as the distance increases the flow has turbulent mixing characteristics before stagnation point and the potential core disappears completely. The presence of the potential core region results in lesser degree of vorticity at stagnation, causing a transition to turbulence in the wall region close to stagnation point. Longer distances result in actual turbulent dissipation and higher heat transfer coefficients at stagnation point but with an average low heat transfer coefficient. Further increasing the impinging distance results in a lack of fluid momentum to drive effective force convection, thus the thermal resistance increases. The appropriate distance increases with increasing Reynolds number of the impinging jet. Shorter distances can cause confinement, resulting in disturbance to the main jet streams. Our study shows that for food freezing applications nozzle-to-food distance ratio should be maintained in the range of 6-10. A wide range of experimental data on heat transfer measurements exists in this regard. Based on heat transfer measurements Gardon and Akfirat (1966) have suggested

an optimal nozzle-to-food ratio of 6-8. The results on heat transfer coefficients and flow visualization studies indicated that heat transfer coefficients depended on nozzle-exit-to-food spacing. This is due to the turbulence induced at the boundary of the jet and the stagnant air. In many impingement applications, this type of turbulence tends to dissipate the energy in the jet region of the flow. To minimize energy dissipation, the nozzle-exit-to-plate spacing may be reduced, but an extremely short spacing between nozzle exit and the food causes unnecessary confinement of the jet. Confinement may prove advantageous in some cases but not so in others. In processing applications where moisture is being removed (baking, drying), confinement is not desirable, but in applications such as thawing, confinement may reduce drip loss by conserving moisture. When the food is positioned downstream of the potential core the freezing air velocity is degraded from its nozzle exit value, the energy of the jet decayed because of the turbulence in free jet region causing lower heat-transfer coefficient and the food is cooled ineffectively. The nozzle-to-plate

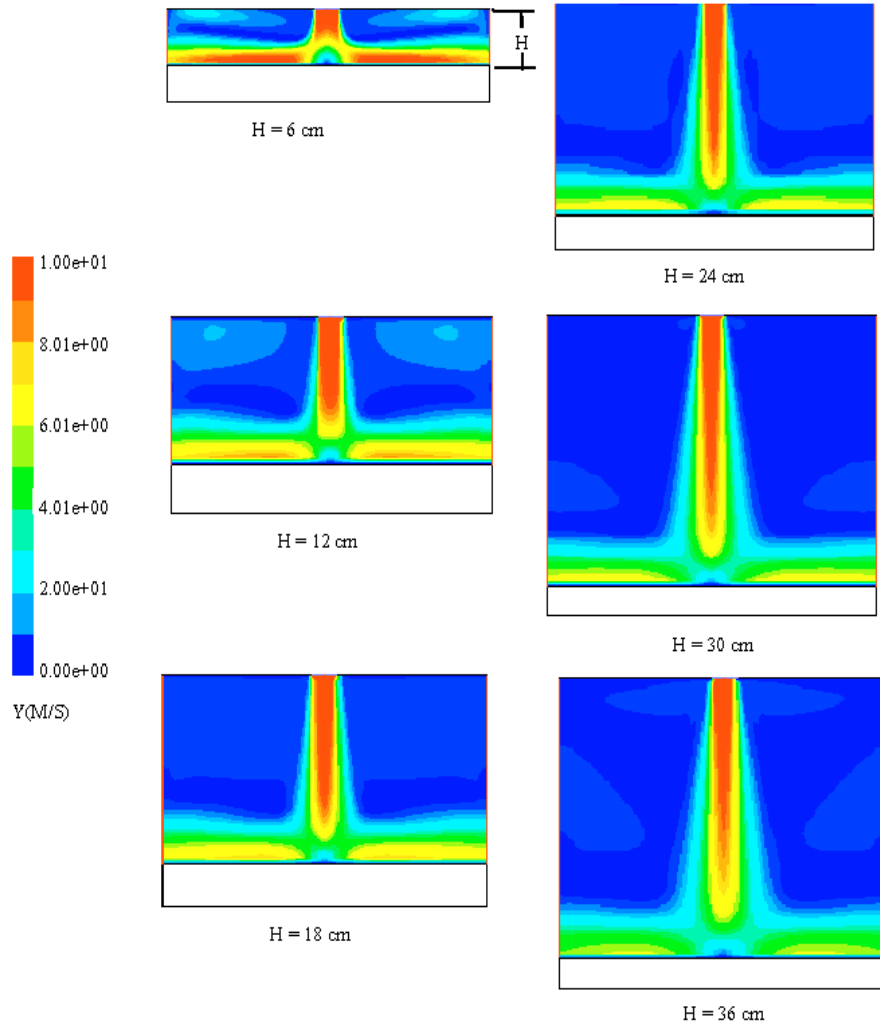


Fig. 10: Effect of nozzle-to-food distance (H) on flow pattern for the  $U_j = 10 \text{ m sec}^{-1}$ ,  $T_j = -10^\circ\text{C}$

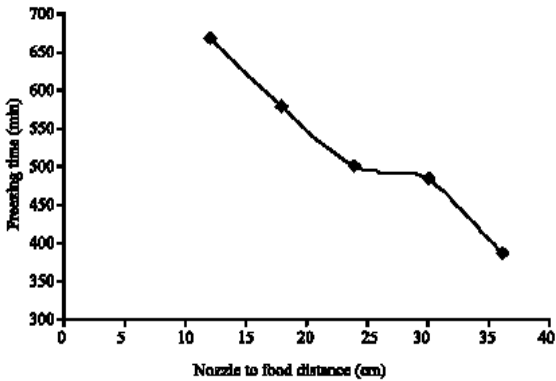


Fig. 11: Influence of nozzle to food distance on freezing time

spacing should be optimized for best results, depending on the application. Such optimization would require predictive relationships for heat transfer coefficients based on the position and flow conditions. Figure 9 also shows the formation of boundary layer on the surface of the product at short distances away from stagnation. Although the boundary layer observed is only the hydrodynamic boundary layer and not a true representation of the thermal boundary layer, it is strongly correlated to the thermal boundary layer and results in a step decline in heat transfer coefficients at short distances away from stagnation.

The heat transfer from a slot jet impinging on a slab shaped food has been studied using CFD (FLUENT6.0) simulations with Reynolds Stress turbulence (RSM)



model. The CFD simulation results show how different impingement conditions (e.g., air velocity and temperature, nozzle to food distance) affect the freezing time of slab shaped foods and the information on these may be used in further optimization studies. The results of this study show that:

- Impinging air temperature is the most important variable, influencing freezing time.

Lowering impinging air temperature, freezing time decreases but beyond critical point, it decreases slowly because the freezing air temperature showed little effect on the heat transfer coefficient.

- The velocity of the impinging air plays an important role in the thermal resistance.

Increasing the velocity consistently diminishes the thermal resistance. Freezing time decreases with increasing impinging air velocity because convective heat transfer coefficient improves at higher velocities.

- The distance between the nozzle exit and the food is the other parameter affecting freezing time. Nozzle-to-food distances in the region of 6-8 are best because they ensure that the potential core is fully decayed and there is no excessive energy dissipation.

These results indicate that with proper optimization of above mentioned parameters, major improvements can be achieved in air impingement food freezing systems.

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#### NOMENCLATURE

$B_j$	=	Width of jet
$d$	=	Thickness of slab shaped food
$H$	=	Nozzle-to-food distance
$h$	=	Convective heat transfer coefficient
$C_j$	=	Specific heat of unfrozen food
$C_s$	=	Specific heat of frozen food
$C_p$	=	Heat capacity at constant pressure
$C_v$	=	Heat capacity at constant volume
$K$	=	Thermal conductivity

$k_j$	=	Thermal conductivity of unfrozen food
$k_s$	=	Thermal conductivity of frozen food
$L$	=	Length of slab shaped food
$L_f$	=	Latent heat of freezing
$P$	=	Pressure
$q_v$	=	Internal source term due freezing
$T$	=	Average temperature
$T_f$	=	Freezing temperature of food
$T_i$	=	Initial temperature of freezing food
$T_j$	=	Temperature of freezing air
$T'$	=	Fluctuating component of temperature
$t$	=	Time
$U$	=	Average velocity
$U_j$	=	Velocity of air at nozzle exit
$u'$	=	Turbulent component of velocity
$\langle u'_i u'_j \rangle$	=	The average value of the fluctuating component of the velocity
$X$	=	Stream wise coordinate
$Y$	=	Coordinates perpendicular to stream wise coordinate direction
$\rho$	=	Density
$\gamma$	=	Kinematics viscosity of the air

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